Machinery Messages

Case History

Remote diagnosis of a pure *Half X* vibration problem in a power plant

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Ballylumford is a 960 MW steam-powered generating plant situated on Larne Lough, Northern Ireland. The facility is over 25 years old and has three 120 MW machines and three 200 MW machines.

This article discusses a problem which occurred on one of the 200 MW turbines. The machine train diagram (Figure 1) shows the general layout. All the shafts are solidly coupled.

After an overhaul in 1993, the High Pressure turbine No. 1 bearing horizontal shaft vibration reading was higher than normal. The 300 μm (12 mils) peak to peak vibration amplitude was all at 1X running speed. However, the machine ran acceptably with this level of vibration for approximately seven months. The shaft relative motion is measured by proximity probes mounted on a carrier ring some 150 mm (6 inches) outboard of the No. 1 bearing centerline. The No. 1 bearing is housed in a pedestal which is free to slide axially as the machine casing expands and contracts.

In March 1993, the machine was shut down over a weekend. No work was performed on the turbine, but on the subsequent startup, vibration levels fluctuated. The horizontal vibration level reached 430 μm (17 mils), with a 1X component of 300 μm (12 mils) and a component at exactly 0.5X running speed of 260 μm (10.2 mils). (Note: individual vibration components do not sum arithmetically to give the overall value.)

A plant engineer phoned Bently Nevada's office in the United Kingdom and faxed plots generated by their Bently Nevada Transient Data Manager® (TDM) System to Bently Nevada's Machinery Diagnostic Services (MDS) group. The computer-based TDM System captures transient vibration data during machine startup and shutdown, and steady-state data during machine online operation.

Common causes of vibration at exactly 0.5X running speed are looseness and rub effects, in conjunction with the presence of a balance resonance (often called "critical speed") at marginally above or below half running speed, respectively. This machine's runup data was not available, so MDS recommended checking the turbine's No. 1 bearing as a first step. The turbine was shut down and the bearing inspected. Its clearance was acceptable.

Upon startup, the same problems occurred, and additional plots were faxed to MDS. The direct (overall) 24- hour vibration trend plot (Figure 3) shows how the vibration levels varied. The three lines (solid, dotted and dashed) on the plot show maximum, average and minimum values, respectively. At a 30 MW load, there was no 0.5X vibration present, and the overall vibration level was 270 μm (10.6 mils). The 1X vibration component increased with increasing load, and at 40 MW, the 0.5X component appeared, at a level of 260 μm (10.2 mils). The 1X vibration component had increased to 400 μm ►

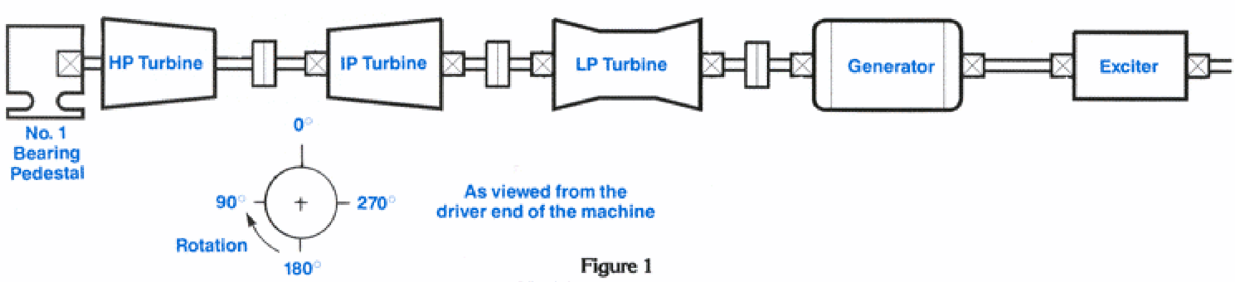


Figure 1
Machine train diagram

(15.7 mils). The two components combined to give overall levels of 450 to 500 μm (18 to 19.7 mils).

As the 1X vibration component level was not excessive when the machine reached running speed but increased with increasing load, unbalance was ruled out as the cause of the high 1X values. Alignment problems were suspected.

The 0.5X component continued to occur as the load was increased to 90 MW. At 90 MW load, the instrumentation showed that the HP pedestal jumped forward on its slides. Coincident with this movement, the 0.5X component disappeared and the vibration level dropped to 400 μm (15.7 mils). The turbine ran with vibration levels of 370 to 400 μm (14.6 to 15.7 mils) for twelve more hours, until an excursion of 20°C (68°F) in steam inlet temperature occurred. The 0.5X component reappeared and the HP vibration reading again increased to 500 μm (19.7 mils).

The 0.5X component remained until further temperature changes occurred. The pedestal moved again, and the 0.5X component disappeared. The vibration decreased to approximately 400 μm (15.7 mils). The turbine ran at this vibration level until it was shut down due to a boiler tube leak. Figure 4 shows the unfiltered orbit and waveforms when the 0.5X component was present.

The No. 1 bearing clearance had been acceptable and no other source of looseness had been established. The 0.5X component was related to temperature and HP pedestal movement. The most likely cause of the 0.5X component was a rotor-to-casing radial rub, as the HP casing became distorted when the HP pedestal stuck on its slides. The orbit's shape indicated that the rub was occurring at the left-hand side of the casing, at the nine o'clock position, as viewed from the driver end of the machine.

Figures 5 and 6 show the HP horizontal vibration and pedestal overall vibration trends from the time the 0.5X vibration disappeared until the machine was shut down.

At approximately 3:00 PM on March 10, the average vibration reading at the No. 1 bearing was approximately 370 μm (14.6 mils), while the pedestal vibration level was 16μm (0.63 mils), a shaft to pedestal displacement ratio of 23 to 1. As it is well-known that this type of bearing pedestal does not have a high mechanical impedance, this ratio was considered to be high. Therefore, it was concluded that only low levels of force were transmitted through the bearing oil film to the pedestal, in spite of the large amplitude of the shaft motion.

It is also noteworthy that the force transmitted to the pedestal increased disproportionately as more of the bearing clearance was taken up by the shaft motion. At 6:00 PM, vibration at the bearing had increased to 390 μm (15.4 mils), a 5.5% increase. However, the pedestal vibration had increased to 24 μm (0.94 mils), a 44% increase. Based on the available information, MDS concluded that the No. 1 bearing was misaligned and was low, partly as a result of settling. Therefore, the bearing was not providing adequate support for the HP shaft.

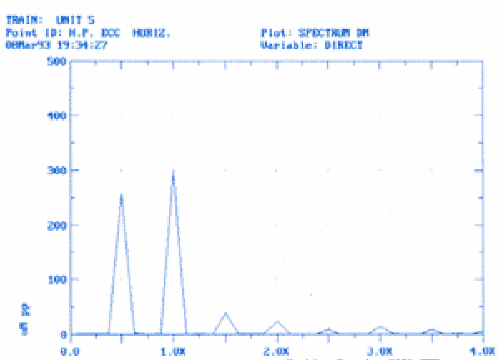


Figure 2
Spectrum plot, showing high vibration levels on the No.1 bearing.

In a phone conversation with the plant engineer, MDS discussed their conclusions regarding both the rub and the No. 1 bearing alignment. The engineer checked the original alignment catenary figures, which confirmed that the No. 1 bearing had been low following the overhaul. During a shutdown over the following weekend, the HP pedestal was raised by 0.5 mm (20 mils). Since the HP casing is attached to the pedestal, the move didn't affect internal alignment between the shaft and the HP casing.

When the turbine was restarted, the rub still occurred for part of the running time, but the underlying 1X vibration level was approximately 20 μm (0.8 mils) lower as a result of the pedestal move (Figures 7 and 8).

An evaluation of this new data did not alter the earlier conclusions. After further discussions with the plant engineer, the turbine was shut down and the following actions were taken:

The HP pedestal was raised 0.5mm (20 mils) more to provide additional load to the bearing.

The top half HP casing steam glands at the No. 1 bearing end were removed and indications of a rub were seen at the nine o'clock position, as had been predicted from the unfiltered orbit plot.

The No. 1 bearing was moved 0.25mm (10 mils) to the right to increase the clearance between the shaft and the left-hand side of the casing.

Conclusion

Following the moves, the turbine ran with overall and 1X vibration levels of 200 μm (7.9 mils) (Figures 7 & 8). The HP pedestal horizontal vibration level was 25 μm (1 mil). No subsynchronous vibration occurred, and the load could be varied as required without causing significant vibration changes. The vertical alignment change reduced the high 1X vibration and the horizontal bearing movement eliminated the 0.5X component. The problems were resolved without sending a vibration specialist to the plant. The on-line data captured and plotted by the TDM System enabled MDS engineers to diagnose the problem from a remote location.

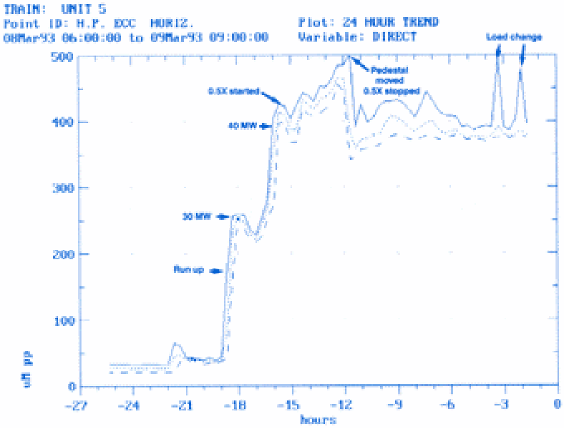


Figure 3
24-hour trend plot of the No.1 bearing overall vibration as load was applied.

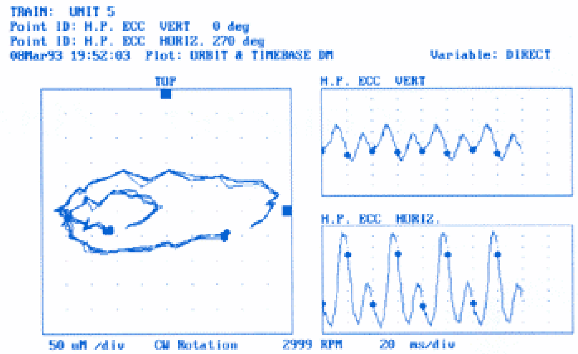
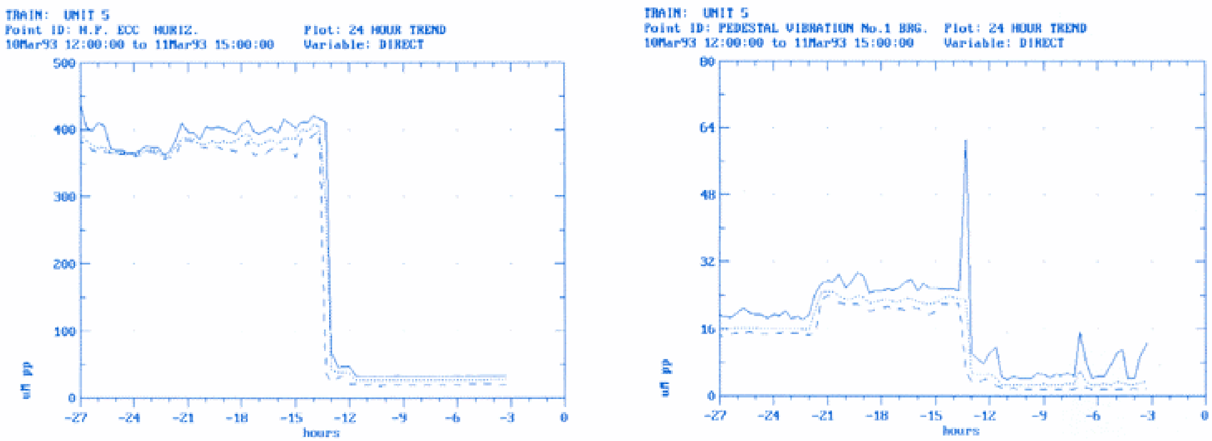
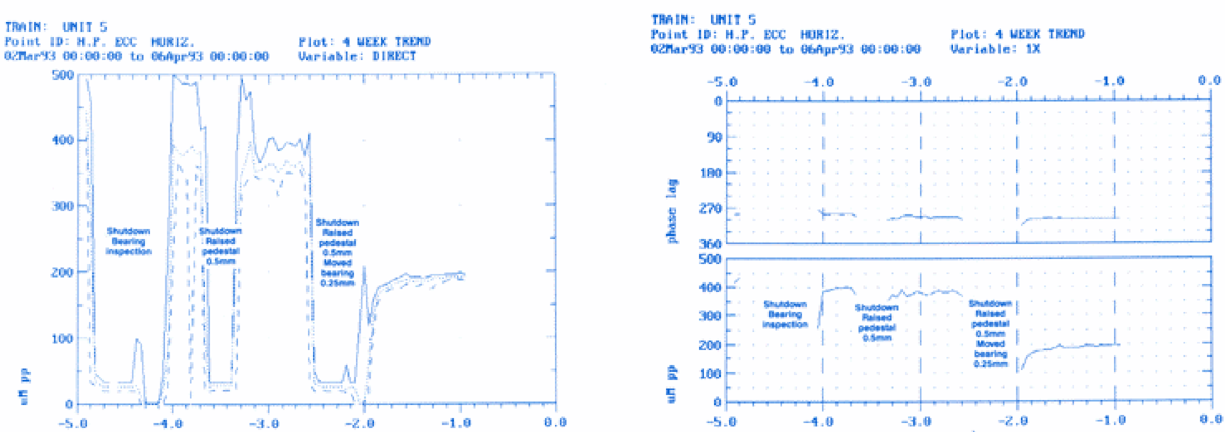


Figure 4
Unfiltered orbit and timebase plot when the 0.5X vibration component was present.



Figures 5 & 6
Shaft and pedestal 24-hour trend plots from when the 0.5X vibration component disappeared to shutdown.



Figures 7 & 8
4-week overall and 1X trend plots, showing lower vibration levels after the change.